HEAT TRANSFER BY IMPINGING JETS ON A MOVING STRIP

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ABSTRACT

This paper deals with heat transfer on a moving plate by means of an impinging jet. Three different turbulence models are used and it turns out that Lam-Bremhorst model is in good agreement with measurements when Re is lower that 5000. In case of moving strip (ratio m $=V_{strip}/V_{jet}$ lower than 1/3), there is almost no effect of m on Nusselt distribution in the stagnation region.

1. INTRODUCTION

Impinging jets are widely used in steelmaking processes (heating or cooling in continuous annealing furnaces for example). The reason is the high efficiency of jets in injecting or removing thermal energy in industrial processes.

Most of the time, it is necessary to identify influential parameters on heat transfer in order to predict the characteristics of a requested device (mean value of exchanged heat flux, temperature homogeneity ...) and to put forward reliable proposals. Predictive methods are based on the use of standard correlations such as $Nu=kPr^nRe^m$ expressing empirical dependence of Nusselt number on Prandtl and Reynolds numbers.

Unfortunately, most of our problems cannot be solved with such correlations :
•local information is needed, but correlations provide global information,
•geometry is specific and no correlation is available in the literature dealing with it,
•the solid can have an important displacement velocity (maximum value is about 15 m/s for typical steelmaking problems) and this case is never considered in literature.

It seems that an attractive way to cope with our problem could consist in using a commercial CFD package. With many other industrial research centers, we decided to launch into numerical simulations with PHOENICS.

The final aim of the work we present here is to numerically quantify the influence of strip displacement on heat transfer coefficient at impinging point. But before that, a lot of intermediate steps have to be cleared :
•we have to check that wall shear stress for both axisymmetric impinging jet and impinging slot jet, are correctly predicted,
•qualitive and quantitative agreement between calculations and measurements must be achieved for local heat transfer, which is far from established according to authors using commercial codes (Morris et al., 1996, Ashforth-Frost et al., 1996).

Much of the discrepancy can be attributed to uncertainties in the turbulence models used, particularly in the low Reynolds number region close to the impingement point. For instance, work by Seyedein et al. (1994) has shown that for impinging slot-jets, low-Re versions of the k-ε model are most suitable and far superior to the standard high-Re model which is the default option for most industrial CFD codes. In particular, the models of Launder and Sharma (1974) and Lam-Bremhorst (1981) have been recommended. Similarly, Polat & Douglas (1990) have shown, that for a confined 2D turbulent air jet impinging on a flat surface, mean properties (pressure and centre-line velocity decay) are quite well predicted independent of near-wall model used, but heat transfer is quite sensitive to near-wall assumptions.

In addition and especially for determining wall damping functions, a very promising trend consists in using direct numerical simulation (DNS) and deriving wall functions for coarse models such as k-ε turbulence model. Nagano et al. at Nagoya Institute of Technology obtained very interesting and general results with this method for both momentum and heat transfer (Abe et al., 1994, 1995, Shimada et al., 1996).

Unfortunately, turbulence models available in commercial packages rarely correspond to the most efficient ones, according to recent literature, to mimic an industrial process characterized by a succession of basic flow situations, Heyerichs et al., 1996. In this paper, we will present different turbulence models included in the standard version of PHOENICS, performing their validation for an impinging jet whenever it is possible.

In addition to near-wall effects, other terms need to be added to the k-ε turbulence model. They have to correct the rate of spread of both the axisymmetric jet and the wall-jet downstream from the impingement point (Rodi, 1980, Malin, 1988 and recently Myszko et al., 1995). These additional corrections have not been implemented in this study.

2. A LIMITATION OF COMMERCIAL CFD PACKAGES

For heat transfer prediction between fluid and solid, most of commercial packages use the well-known Jayatillaka's correlation which expresses Stanton number :

$$St = \frac{s}{Pr_t\left(1 + P\sqrt{s}\right)}$$

with Pr_t : turbulent Prandtl number

$$P = 9\left(\frac{Pr}{Pr_t} - 1\right)\left(\frac{Pr_t}{Pr}\right)^{0,25}$$

$$St = \frac{q_w}{\rho c_p U_p (T_p - T_w)}$$

where T_w : wall temperature

T_p : temperature at node P close to the wall.

$$s = \frac{1}{V_p^{+2}} = \frac{1}{2} C_f \qquad \text{with :}$$

$$V_p^+ = V_p/u^*,\ u^* = \sqrt{\tau_p/\rho} \text{ and } \tau_p = \mu\left(\frac{\partial V}{\partial y}\right)_{y=0},$$

C_f is the friction coefficient and V is the tangential velocity.

We finally obtain for heat flux density :

$$q_w = \lambda_f \frac{\Delta T}{\Delta y} \frac{Pr\, y^+}{Pr_t\left(V_p^+ + P\right)} \qquad \text{and}$$

$$h = \frac{\lambda_f}{\Delta y} \frac{Pr\, y^+}{Pr_t\left(V_p^+ + P\right)} \quad \text{where } y^+ = y\, u^*/\nu$$

In the logarithmic sub-layer, $V^+ = \frac{\ln Ey^+}{\kappa}$

(where κ is the Karman constant), so :

$$q_w = \lambda_f \frac{\Delta T}{\Delta y} \frac{Pr\, y^+}{Pr_t\left(\frac{\ln Ey^+}{\kappa} + P\right)}$$

$$\text{or } h = \frac{\rho c_p u^*}{Pr_t\left(\frac{\ln Ey^+}{\kappa} + P\right)} \quad \text{for } 30 \le y^+ \le 500.$$

That means that for an impinging slot jet, for instance, heat transfer should be zero at the axis, because wall shear stress is zero at impinging point. But experimental results show that maximum heat flux is transmitted in the impinging region of the jet, and it turns out that we absolutely have to adapt standard modelling in the case of heat transfer by means of impinging jets.

It is possible to solve this problem making the choice to have another reference velocity than skin friction velocity u^*. Chieng and Launder, 1980, proposed to use the square root of turbulent kinetic energy at the edge of viscous sublayer as a reference velocity. But for some cases, it remains important to still use u^* as the reference velocity and there is no rule to switch from one reference to the other.

In order to get rid of correlation for heat flux, we decided to use the several low-Reynolds turbulence models. Provided that the first node in the fluid side is located in conductive thermal sublayer, heat flux is obtained directly from Fourier's law :

$$q_w = -\lambda_f \overrightarrow{\text{grad}T} \cdot \vec{n}$$

where λ is the thermal conductivity of fluid and $\vec{n}$ is the external normal vector to the wall.

Main aim of this paper will be devoted to test numerical predictions obtained with different low-Reynolds turbulence models included in standard version of PHOENICS 2.1.3 (Lam-Bremhorst, Lam-Bremhorst with Yap correction and Chen-Kim) by means of a comparison with experimental results.

3. THE TURBULENCE MODELS USED

There are two approaches to the modelling of near-wall regions:

(i) As in the original paper of Launder and Spalding (1974), use the standard high-Re k-ε model coupled to a wall-function to "jump over" the near wall region. This approach has the advantage of economy, but pre-supposes the structure of the wall layer.
(ii) Use a low-Re k-ε model which by correctly reflecting the effect of the wall on velocity fluctuations and by including the viscous influence on dissipation should lead to an improved result. The penalty will then be the cost of a much finer grid and possibly a less stable simulation.

This paper shows the application of three low-Re versions of the k-ε model which are available as built-in options in PHOENICS, and have been recommended by other authors (Patel(1984), Seyedein (1994)] as particularly useful for impinging jets with heat transfer. These are the Lam-Bremhorst model (LB), the same model with the correction by Yap (1987), (LBY) and the model by Chen and Kim (1987) (CK).

The k-ε model conservation equations are :

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i}\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i} + P_k - \rho\varepsilon \quad (1)$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i}\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i} + \frac{\varepsilon}{k}(f_1 c_{1\varepsilon} P_k - f_2 c_{2\varepsilon}\rho\varepsilon) + Y \quad (2)$$

where :

$$P_k = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \qquad \mu_t = \frac{f_\mu c_\mu \rho k^2}{\varepsilon}$$

In the high Reynolds number version the empirical coefficients in the model are :

$c_\mu=0.09, c_{1\varepsilon}=1.44, c_{2\varepsilon}=1.92, \sigma_k=1.0, \sigma_\varepsilon=1.3$

The multipliers appearing in front of the first three coefficients in the equations (1) and (2) are equal to unity, i.e, $f_1=f_2=f_\mu=1.0$.

In the LB model :

$$f_1=1+\left(\frac{0.05}{f_\mu}\right)^3,$$

$$f_\mu=(1-\exp(-0.0165 Re_n))^2\left(1+\frac{20.5}{Re_t}\right)$$

$$f_2=1-\exp(-Re_t^2),\ Re_t=\frac{\rho k^2}{\mu\varepsilon},\ Re_n=\frac{\rho\sqrt{k}n}{\mu}$$

where n is the normal distance to the nearest wall and $k=\partial\varepsilon/\partial n=0$ at the wall (see Patel et al., 1986).

These additional terms introduce damping of turbulence near the wall without the need for extra sources.

The LBY model

In flows nearing separation, and especially when the dissipation equation is integrated to the wall, the predicted values of the turbulent length scale tend to be excessive. The last term, Y, in the ε equation is then an additional volumetric source which removes this deficiency. Thus :

$$Y=\max\left(0.83\rho\,(L/L_e-1)(L/L_e)^2\varepsilon^2, 0\right)$$

$$L=k^{1.5}/\varepsilon,\ L_e=c_L n,\ c_L=c_\mu^{0.75}/\kappa\ (=2.495)$$

The CK model

In recognition of the multi-scale nature of turbulence, Chen and Kim, 1987, proposed the addition of a second time scale which is based on the production rate of k, (k/P_k) to the usual time scale,(k/ε). This was done to improve the dynamic response of the ε equation.

Implementation of the model involves in essence the addition of an extra source term, S_ε , in the equation, to represent the energy transfer rate from large to small eddies :

$$S_\varepsilon=\rho f_1 c_{3\varepsilon} P_k^2/k.$$

The net effect is to increase ε (hence decrease k) when the mean strain is strong($P_k/\varepsilon > 1$) and conversely decrease ε when the mean strain is weak ($P_k/\varepsilon < 1$). This model is used in conjunction with the LB model for non-equilibrium low-Re applications. A new constant has been introduced, $c_{3\varepsilon}=0.25$, and some of the standard k-ε model constants are modified as follows :

$c_{1\varepsilon}=1.15,\ c_{2\varepsilon}=1.9,\ \sigma_k=0.75,\ \sigma_\varepsilon=1.15$

4. TURBULENT PRANDTL NUMBER

Transport equation for enthalpy can be expressed in the following way for mean temperature T (constant thermal properties) :

$$\frac{\partial T}{\partial t} + u_j\frac{\partial T}{\partial x_j} = -\overline{u'_i u'_j}\frac{\partial u_i}{\partial x_j} + \frac{\partial}{\partial x_j}\left(a\frac{\partial T}{\partial x_j} - \overline{u'_j T'}\right)$$

where a is the thermal diffusivity : $a = \frac{\lambda}{\rho c_p}$.

Rather than solving a transport equation for $\overline{u'_j T'}$, the Boussinesq closure assumption can be made :

$$\overline{u'_j T'} = -a_t\frac{\partial T}{\partial x_j}.$$

The remaining problem consists in determining the turbulent diffusivity a_t and a turbulent Prandtl number is introduced : $Pr_t=\frac{\nu_t}{a_t}$. A lot of studies have been devoted to the assessment of Pr_t (see Launder and Spalding, 1972). It turns out that far from boundary layers, $Pr_t\approx0.9$. Inside boundary layers, some relations are available in the litterature in order to get Pr_t. Roughly speaking, we observed two categories of analytical expressions : ones which directly include distance to the wall (first category) and other that are derived from local turbulent properties (second category).

first category example :

For Pr=0.7 (air), Xia and Taylor, 1993, proposed :

•Pr_t	=	0.7
		$y^+ \leq 5$
•Pr_t	=	$1.4 - 0.7(13-y^+)/8$
		$5 < y^+ \leq 13$
•Pr_t	=	1.4
		$13 < y^+ \leq 17$
•Pr_t	=	$0.95 + 0.45(25-y^+)/8$
		$17 < y^+ \leq 25$
•Pr_t	=	0.95
		$y^+ > 25$

second category example :

Morris et al. recommended in their paper the following expression :

$$Pr_t = 0.225 \left(\frac{5.4 + P_k/\varepsilon}{1.2+P_k/\varepsilon}\right)$$

Dispersion for Pr_t is generally important but most of the time Pr_t is contained between 0.7 and 1.5. For this preliminary stage of our study, we fixed Pr_t to a constant value : Pr_t=0.95.

5. NUMERICAL PROCEDURE

Hybrid discretization scheme was used for all variables. Because we have difficulty in converging for ε, wholefield solution method was prefered for this variable rather than slab by slab method. We observed that Lam-Bremhorst low-Reynolds turbulence model was easier to converge than other models.

In order to facilitate convergence, relaxation for velocity components and ε was intensified near the impinged plate. Typical thickness of cells adjacent to the plate is about 10 µm. This refined mesh allows grid independent results. The cost is the great number of cartesian cells : about 270000 for 3D configurations.

Calculations were performed on a node of IBM SP2 computer. About 1500 iterations were required to get converged results in about 15 CPU hours.

6. NUSSELT NUMBER CALCULATION

There are a lot of possibilities to express Nusselt number, depending on the reference temperature. Here, we decided that reference temperature is temperature T_{jet} at nozzle jet exit .

Let h be the heat transfer coefficient (see Figure 1) :

$$\lambda_f(T_F-T_w)/y_f = h(T_{jet}-T_w) \quad \text{so } h=\frac{\lambda_f}{y_f}\frac{(T_F-T_w)}{(T_{jet}-T_w)}$$

with T_w : wall temperature
T_F : fluid temperature at point F

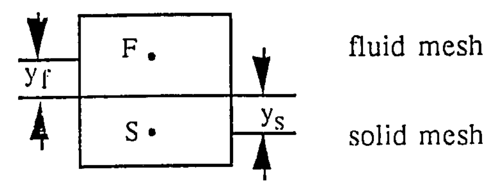

Fig 1. Main parameters for heat transfer calculation

if Nu =hD/λ_f, then we have :

$$Nu = \frac{D}{y_f}\frac{(T_F-T_w)}{(T_{jet}-T_w)}.$$

All parameters are known, except T_w. In order to get T_w, we express q_w in two different ways :

$$q_w = \frac{1}{\left(\frac{y_f}{\lambda_f}+\frac{y_s}{\lambda_s}\right)}(T_F-T_S) = \frac{\lambda_f}{y_f}(T_F-T_w)$$

T_S : solid temperature at point S

$$\text{so } T_w = T_F + \frac{y_f}{\lambda_f}\frac{1}{\left(\frac{y_f}{\lambda_f}+\frac{y_s}{\lambda_s}\right)}(T_S-T_F).$$

7. SHEAR STRESS COMPARISON FOR AXISYMMETRIC IMPINGING JETS

The first stage of validation concerns shear stress comparison in the vicinity of the stagnation point. Because shear stress measurement is extremely difficult to realize, we always have to worry about the reliability of experimental results : it is important to have non intrusive sytsem and validation of the measurement system itself is required.

For shear stress measurement in water, the use of polarographic method (Lebouché and Martin, 1975) is probably the most efficient and reliable system : this electrodiffusion method requires to have a probe embedded in the wall and does not disturb the flow.

Flow structure information near stagnation point is scarce. Aglat et al., 1996, determined local friction and mass transfer by means of a polarographic probe in case of an array of round jets. Recently, Alekseenko and Markovich, 1996, published precise wall shear stress measurements for a single round jet near the stagnation region. The experimental set-up consists of a rectangular vertical channel ; a round nozzle (D=10 mm) is inserted in the channel containing an aqueous solution and a jet is issuing from the nozzle.

The three low-Reynolds turbulence models previously described were used to simulate the configuration experimentally investigated by Alekseenko et al. Comparison between numerical and experimental is shown in Figure 2.

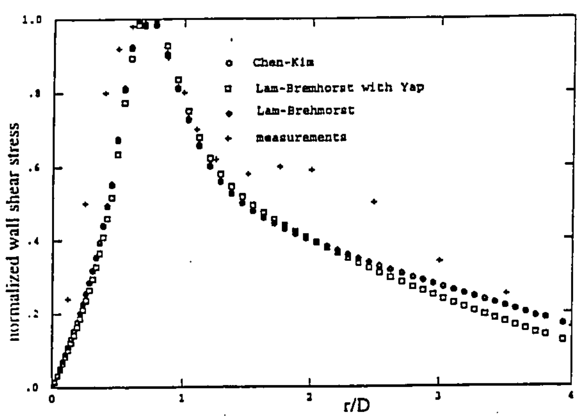

Fig 2. Wall shear stress comparison, Re=24800, H/D=3

The following points can be observed :

•there is a qualitative agreement between measurements and calculations : important increase of wall shear stress followed by a progressive decrease ; it is satisfactory to notice that maximum shear stress is located at about r/D=0.7 for both measurements and calculations,

•all the three turbulence model fail to predict the plateau located at about r/D=2 ; Heyerichs and Pollard, 1996, already noticed that accuracy of different low-Reynolds turbulence models was quite different for heat transfer prediction, and that the k-ω turbulence model of Wilcox was probably the best one to quantitatively predict such a plateau in confined geometrics, Heyerichs et al., 1996 ; unfortunately, this model is not included in standard version of PHOENICS,

•although Yap correction is often recommended in case of impinging jet (see Craft et al., 1993, for the influence of Yap correction on Nusselt number), the disagreement is more pronounced for LBY than for the other models ; but we have to keep in mind that all results are normalized by maximum wall shear stress and, for asbolute values, the trend can be different,

•LB and CK give simular results.

Comparison was also performed for lower Reynods number (Re=6300, Figure 3) using Lam-Bremhorst turbulence model. For this configuration with no plateau, an excellent agreement is obtained. This indicates that Lam-Bremhorst is excellent when there is a regular decrease of wall jet velocity.

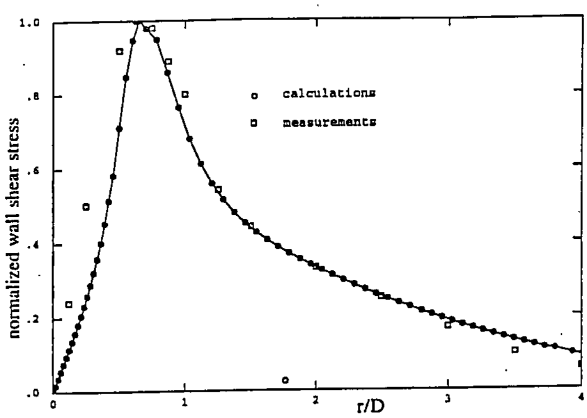

Fig 3. Wall shear stress comparison, Re=6300, H/D=3

8. HEAT TRANSFER COMPARISON FOR AXISYMMETRIC IMPINGING JETS

The second stage of validation deals with heat transfer comparison near stagnation region. Measurements performed by Lee et al. were considered : an axisymmetric air jet is issuing from a long pipe and impinges an electrically heated plate. Measurements were carried out along a radial position, by means of a liquid crystal system ; effect of Reynolds number and H/D ratio was experimentally studied. For a fixed H/D value, authors found that Nu/Re^n was independant of Re in stagnation region :

n	0.52	0.52	0.58	0.70
H/D	2	4	6	10

For the numerical simulation, Lam-Bremhorst turbulence model was selected.

The comparison was performed for H/D=4, Figure 4.

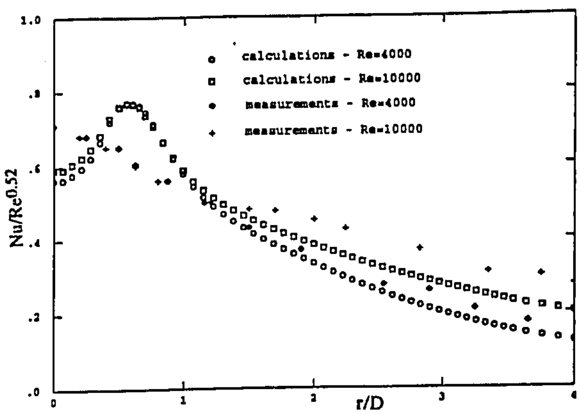

Fig 4. Local Nusselt number comparison, H/D=4

Mean Nu/Re^n value from r/D=0 to r/D=1 is about 0.65 ; this is in agreement with measurements.

The major difference is that there is a local minimum for Nu at stagnation point, and a maximum corresponding to the edge of the jet (r/D ≈ 0.6). This was already experimentally observed by some authors (Popiel et al., 1980, Re=1820) and does not appear in Lee's measurements : it would be interesting that measurements with Lee's technique are compared with results supplied by other measurement systems to assess coherence of different experiments.

Another difference is that measurements exhibit an important effect of Reynolds number starting from r/D = 2. According to calculations, increase of Re leads to only a relatively small increase for Nu/Re^n, whereas the gap is more pronounced for measurements : there is an underprediction of Nu/Re^n for Re=10000 and a correct agreement for Re=4000.

For H/D=6 (Figure 5), the same trend as for H/D=4 is observed and the discrepancy has the same order of magnitue than for H/D=4. In the stagnation region, agreement between measurements and calculations is correct, but predicted decrease of Nu is still too rapid.

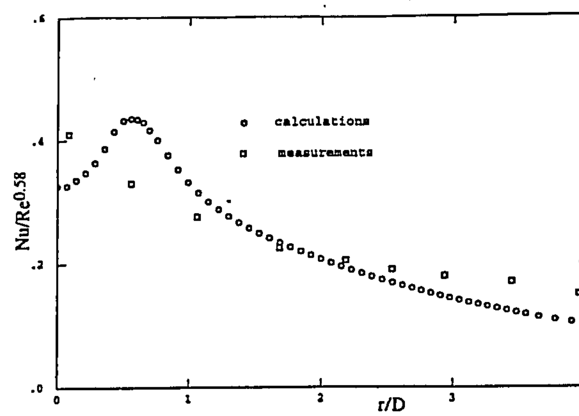

Fig 5. Local Nusselt number comparison, H/D=6, Re=10000

9. INFLUENCE OF STRIP VELOCITY ON HEAT TRANSFER

In industrial problems, jet impinges on a moving plate whose maximum velocity is about 15 m/s (in a reheating furnace for instance). As far as we know, there are no experimental results for such a configuration. Then, two main questions arise :
1°/ is there any influence of the strip velocity on heat transfer in the stagnation region ?
2°/ does the strip displacement can affect the mean heat transfer coefficient and is it possible to still make use of correlations obtained with a static solid ?

Because we believe that previous stages testify to the reliability of numerical simulation to predict conjugate heat transfer, PHOENICS was used. A schematic view of the configuration appears on figure 6.

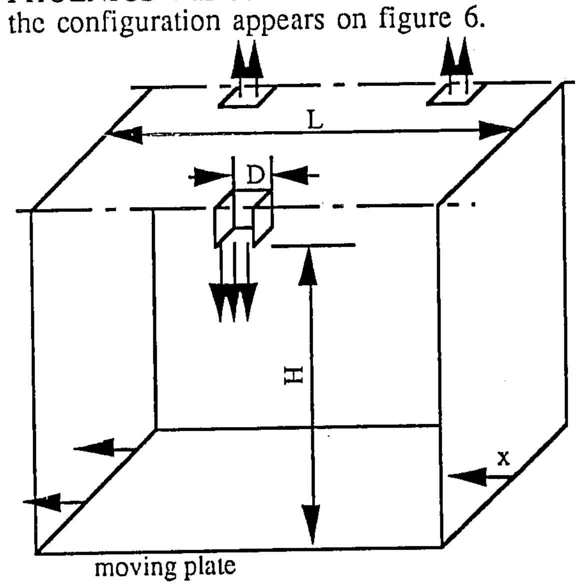

Fig 6. Schematic view of the simulated geometry

The geometry is confined and main parameters are expressed in the following table :

turb. model	Re	Pr	D	H/D	L/D
Lam. Bremh.	10500	0.7	1 cm	7	12

Three different strip velocities were investigated ($m = V_{strip}/V_{jet}$ = 0, 1/6, 1/3). Nusselt number evolution is represented on Figure 7. The most interesting point is that there is no influence of m, within the studied range of parameters, on Nusselt number in the stagnation region. Upstream of the impingement point, a slight decrease of Nusselt number is predicted, whereas there is an increase downstream. This should be confirmed, because the upstream location corresponds to wall jet and strip having opposite direction and an increase of heat transfer would be expected.

High value of Nu is predicted for x=0 when m>0 : this is associated with the leading edge of a boundary layer.

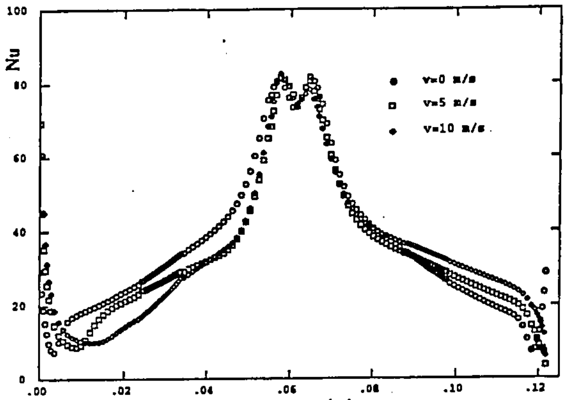

Fig 7. Nusselt evolution for different strip velocities

10. CONCLUSION

Main interesting aspects of this study are summarized in the following three points :

•use of low-Reynolds turbulence model provides a good assessment of heat transfer in stagnation region for jet impingement on a plate,

•for wall jet outside the stagnation point, discrepancy between measurements and calculations is all the more important as Reynolds number is high : good agreement for both heat transfer and wall shear stress when Re is about 5000 and important disagreement when Re is higher than 10000,

•within the studied range of parameters, there is a minor influence of ratio V_{strip}/V_{jet} on heat transfer near stagnation region.

NOMENCLATURE

D : jet nozzle diameter [m]
h : heat transfer coefficient [$Wm^{-2}K^{-1}$]
H : nozzle to plate spacing [m]
k : turbulent kinetic energy [m^2s^{-2}]
m : velocity ratio (V_{strip}/V_{jet})
$Nu = hD/\lambda$: Nusselt number
Pr : Prandtl number
Pr_t: turbulent Prandtl number
q_w : heat flux density [Wm^{-2}]
P_k : mean turbulence generation rate [m^2s^{-3}]
r : radial position from stagnation point
$Re = V_{jet}D/\nu$: Reynolds number
u^* : friction velocity [ms^{-1}]
u_i : component i of mean velocity [ms^{-1}]
u_i' : fluctuating velocity
T : mean température [K]
T' : fluctuating temperature [K]
V_{jet} : jet velocity at nozzle exit [ms^{-1}]
y^+: normalized wall distance

Greek symbols
λ : thermal conductivity [$W\,m^{-1}K^{-1}$]
ε : dissipation rate of k [m^2s^{-3}]
μ : dynamic viscosity [Pa.s]
ν : kinematic viscosity [m^2s^{-1}]

Subscripts
F : first node in fluid side
S : first node in solid side
f : fluid
s : solid
w : wall

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